HEAT TRANSFER COEFFICIENTS OF STEAM WATER MIXTURES

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by

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PREFACE

It has become apparent during the course of instruction at the Naval Postgraduate School that the phenomena of heat transfer is of ever increasing importance in many fields of engineering. The attainment of higher thermal efficiencies ultimately becomes a problem in heat transfer. The concept of the heat transfer coefficient has become an accepted method for predicting the relative ease with which heat flow will occur.

In conversations with Professor E. E. Drucker, of the U. S. Naval Postgraduate School, the authors were made aware of the lack of adequate data on transfer coefficients as they relate to wet steam. Such lack of data was considered sufficiently important to be investigated.

The experimental work of this thesis was conducted at the U. S. Naval Postgraduate School from March to June of 1953.

The authors desire to express their appreciation to Professor E. E. Drucker for his extensive assistance in the preparation of this thesis, to the Research Division of the Babcock and Wilcox Company for their invaluable aid in the instrumentation of the test section, and also to Chief Jones, USN, and Chief Wallace, USN, for their aid in the construction of the experimental set-up.

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LIST OF ILLUSTRATIONS

TABLE OF SYMBOLS AND ABBREVIATIONS

A	Area of heat transfer surface, square feet.
c_p	Specific heat at constant pressure, Btu/(lb)(deg F).
d	Diameter, feet.
G	Mass velocity, lb/(hr)(sq ft of cross section).
h	Coefficient of heat transfer between fluid and surface, Btu/(hr)(sq ft)(deg F).
k	Coefficient of thermal conductivity, Btu/(hr)(ft)(deg F).
L	Heated length of tube, feet.
M	Mass flow rate, lb/hr.
u	Fluid viscosity, lb/(hr)(ft).
P	Total pressure, lb/sq in.
π	3.1416....
ρ	Fluid density, lb/cu ft.
q	Rate of heat transfer, Btu/hr.
r	Radius, feet.
T	Temperature, degrees Fahrenheit.
v	Fluid velocity, ft/hr
x	Moisture content of steam, lb water/lb mixture, percent.
Nu	$\frac{hd}{k}$, Nusselt Group, dimensionless.
Pr	$\frac{uc_p}{k}$, Prandtl Group, dimensionless.
Re	$\frac{\rho Vd}{u}$, Reynolds, Group, dimensionless.

Subscripts:

1	inlet.	w	water.	t	tube.	d	depth
2	outlet.	s	steam.	o	outer.	i	inner

SUMMARY

The object of the development and preparation of this thesis was to determine the local or point coefficients of heat transfer to wet steam at moderate pressures, and to note their manner of variation with respect to moisture content of the steam. To accomplish this, steam at a known condition was passed through a vertical steel tube of 0.50 inches inside diameter which was heated electrically.

Coefficients were determined under the following conditions: moisture content in the range from 0 to 6 percent, absolute pressures up to 215 psia, and a variation in mass velocity from 73,000 to 172,000 pounds per hour per square foot.

The effects of moisture content were found to be significant, causing a rapid increase in the coefficient with increasing percent moisture. The local coefficients as they are herein presented vary from 110 BTU's/hr-Ft^2-$^\circ$F for saturated steam to values approaching 6500 BTU's/hr-Ft^2-$^\circ$F for steam with a moisture content of about 6%.

INTRODUCTION

Steam power has been long established as the primary means of ship propulsion in the U. S. Navy. In the advent of atomic power for naval vessels, it is entirely probable that steam will continue to be the means by which this power is transmitted to the main engines.

The Navy's interest in the improvement of the thermal efficiencies of boilers has led to the utilization of higher temperatures and pressures. It is felt that additional improvement may be brought about by critical re-evaluation of the current concept of boiler design as it relates to heat transfer. Consequently, it is important to know as much as possible about the mechanism of heat flow to the working medium as it passes through successive stages from saturated water to saturated steam. One aspect of this problem may be solved by experimentally determining the coefficients of heat transfer to steam with varying degrees of moisture content.

In this thesis work, experimentation was conducted in an effort to determine the coefficients for steam with a quality of 96 to 100 percent. Although this is a fairly narrow range of moisture content, the results did tend to indicate the trend with which the transfer coefficients varied with quality. Time was a limiting factor in preventing further investigation of steam with higher moisture content.

Heat transfer coefficients of this kind, that is, coefficients relating to forced convection, are usually correlated with respect to the local Nusselt, Reynolds, and Prandtl numbers. Such correlation, however, is invariably made for fluids which are of a single phase only. Obviously this was not the case in the present investigation. In addition, a thorough survey of all existing literature failed to yield any data on values of density, viscosity, or thermal conductivity of wet steam, thus preventing the computation of the above dimensionless quantities. Further research to determine some of these parameters would be extremely useful.

This work may be considered as an extension of the investigation of heat transfer in two phase flow. It represents some of the initial data available for steam at relatively low flow rates and low heat flux densities.

CHAPTER I

REVIEW OF LITERATURE ON TWO PHASE FLOW

A survey of the broad field of heat transfer indicates that a large quantity of data has been collected on the heat flow to fluids inside tubes. Unfortunately very little data has been published with respect to gas-liquid mixtures.

The lack of information on temperature and velocity distributions has made it necessary to correlate experimental data on heat transfer coefficients by means of empirical equations in terms of certain dimensionless groups. Perhaps the most highly utilized equation is the Nusselt relation for single phase flow in forced convection through a tube (4):---

$$Nu=0.024\ Re^{0.8}\ Pr^{0.4} \qquad (1)$$

It has been found by McAdams and others (4), (5) from data for vaporization inside tubes that the local film coefficients, in the preheat section where the fluid is initially heated, are usually somewhat higher than would be predicted from the Nusselt relation. This result is logical from the standpoint that some vapor bubbles may have formed in the preheating section, with the possibility that they disrupt the film locally as they are formed on the tube wall. As they move out into the fluid they would condense thus releasing their heat of condensation. The combined effect is an increase in heat transfer coefficient. In a comparable manner, it might be expected that as far as the influence on

the film coefficient is concerned, the coefficient will increase when gas passes through a system simultaneously with a liquid.

In 1949 Bergelin(2) observed that if a gas and a liquid pass simultaneously in upward flow through a tube, as the gas-liquid ratio is increased, the following three flow conditions could be distinguished:

(a) bubble flow, when gas bubbles pass individually through the tube;

(b) slug flow, when alternatively slugs of gas and liquid pass through the tube, and

(c) annular flow, when the liquid flows in an annulus along the tube wall, while the gas passes at a much higher velocity through the center of the tube. Verschoor and Stemerding(8) in their investigation of the transfer coefficients to air and water mixtures verified the observations of Bergelin. They found that as their air to water mixture was increased, the heat transfer coefficient increased until a maximum was reached that coincided with the transition from slug to annular flow. Prior to reaching this maximum there was a change in slope in their curve which corresponded to the change from bubble flow to slug flow. It is interesting to note that their heat transfer coefficients increased with an increase in flow density, however this was of secondary importance compared to the influence of varying the air-water mixture.

Yoder and Dodge(9) measured film coefficients of Freon-12 boiling in a vertical tube. Their results showed that pressure and flow rate had very little effect on the coefficients, but as the relative amount of vapor increased from 40 to 100 percent, there was a very significant decrease in the heat transfer coefficients.

CHAPTER II

EQUIPMENT

The equipment was a flow system in which steam was taken from a main line and passed through a throttle valve, a cooler, the test section, a condenser, and to a weigh tank. The assembly is shown schematically in Figure 1. The steam was supplied from a Babcock and Wilcox FM boiler at a pressure of 200 psig. After passing through a throttle valve the steam was made to flow through a cooler which was intended for use as a method of varying the quality. It was never subsequently used since time limited the investigation to a percent moisture not greater than 6 percent. On either end of the test section an Ellison combination throttling and separating calorimeter was installed for quality determination. Prior to the condenser an orifice of 0.166 inches diameter was installed to maintain the pressure in the test section. A bypass was provided around the orifice to permit adjustment of the flow rate independently of pressure.

The test section as seen in Figures 2 and 4 consisted of a vertical steel tube having an inside diameter of 0.50 inches and an outside diameter of 1.25 inches. Its overall length was approximately 27½ inches, but with an active length of 24 inches for purposes of heat transfer. Recesses were cut in either end of the test section to minimize axial heat loss. The thermocouples in the test section were made of No. 28 gauge chromel-alumel wire, and were installed by the Research

Division of the Babcock and Wilcox Company. Four pairs of thermocouples were installed along the tube, one of each pair at the outer surface of the tube, the other at a known depth below the surface thermocouples. The leads of the thermocouples were threaded through a milled slot and out a $\frac{1}{4}$" tube welded to the test section. The original thought was that the thermocouples as located would permit determination of the temperature gradient across the wall of the tube, thus the inside surface temperature could be found by extrapolation.

At either end of the test section, thermocouple probes were projected into the fluid stream for measuring the temperature of the steam. These probes were obtained from Leeds and Northrop, and manufactured of iron-constantan wire.

The heating element consisted of four independent heaters, each containing 29.8 feet of No. 18 Nichrome V wire. The wire was wound around the tube in a single layer. Each heater was approximately 6 inches in length. In order to prevent shorting of the heater coils, the wire was insulated with glass sleeving, and a thin layer of mica, 0.0015 inches thick, was wrapped around the tube before the heaters were wound. Power supply to the heaters was controlled by four 115v-15amp Powerstats. See Figure 3. The total power to each heater was measured by portable wattmeters.

The insulation of the test section was accomplished by winding the tube, outside the heaters, with strips of 1/8"

[illegible]

[illegible]

[illegible]

[illegible]

thick asbestos cloth. This was followed by applying a length of standard $1\frac{1}{2}$" magnesia pipe insulation. The entire section was then covered with an aluminum sheet casing. Provisions were made so that air, heated to approximately the same temperature as the outer surface of the insulation, could be supplied to the space between the insulation and casing, thus minimizing any heat loss due to convection, and reducing the conducted heat loss through the insulation.

CHAPTER III

OPERATING PROCEDURE

There were four primary variables associated with the apparatus: heat flux, flow rate, quality, and pressure. It was found, however, during preliminary experimentation, that the minimum moisture content of the steam available for use was approximately 6%. Interest at this time was primarily centered on the region between 0 and 10 percent moisture, thus necessitating throttling the steam to lower pressures in order to obtain a given quality at the entrance of the test section. Such an arrangement meant that only the heat flux, pressure, and flow rate could be varied independently of one another. The only other limiting factor associated with the equipment was, that due to the design of the heaters and the power supply used, the maximum heat input was restricted to 1 kw per heater, or a total of 4 kw.

The procedure was to select an approximate quality at the entrance of the test section by setting the throttle valve accordingly. The flow rate was either taken as that provided by the pressure across the orifice in the line to the condenser or adjusted by means of the by pass around the orifice. Power was then supplied to the heaters and the system was allowed to come to a steady-state condition.

The approach to steady-state condition was determined by noting the change in the millivolt reading of the thermocouples contained in the test section. When they became

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

approximately constant, steady-state was considered to exist. It normally took from 1/2 to 3/4 hours before the system reached equilibrium. It was never possible to keep temperatures constant over a period longer than a few minutes, even though heat input and flow rate appeared unchanged. Each run was made at constant flow rate, heat flux, pressure and quality of the inlet steam.

For each run, temperatures, pressures, flow rate, and power input were recorded. Data points were taken at various settings of power inputs, flow rates, and pressures. Most of the runs were concluded at a power input such that the minimum change in quality which occurred through the test section was 3%.

CHAPTER IV

METHOD OF CALCULATION

The fundamental law of heat transfer by convection may be stated by the mathematical expression:(4)

$$q = hA\,\Delta T \tag{2}$$

The factor of proportionality h, defined in the equation, is called "the film coefficient of heat transfer." It is this coefficient which was computed.

For each set of conditions imposed on the test section, the density of heat flux, the inside surface temperature of the tube, the ΔT, the quality of the steam, and the local coefficient of heat transfer were calculated. Each of these factors was evaluated at four positions corresponding to the locations of the thermocouples inside the tube wall.

The power dissipated in the heating elements was obtained from wattmeter measurements. The heat flux density was then calculated from the power dissipated and the dimensions of the heat transfer surface.

Early in the process of collecting data for the investigation, it was discovered that the thermocouple pairs in the test section were indicating unreasonable temperature gradients through the tube wall. It became necessary to calculate the inner surface temperature by applying to the depth thermocouple a computed temperature drop corresponding to the heat input. The temperature drop was calculated using

the Fourier conduction equation for steady-state heat transmission:(4)

$$q = -kA\frac{dt}{dx} \tag{3}$$

For a circular tube, this equation may be integrated to give:

$$q = \frac{2\pi k L \Delta T}{\ln r_o/r_i} \tag{4}$$

The depth thermocouples were shielded from any direct influence of the heaters, and thus were considered to be more accurate than those on the surface of the tube.

The temperature of the steam was measured at both the entrance and exit of the test section, however, these measurements were consistently erratic and failed to give desired results. It was felt that greater reliability would result from assuming the saturation temperature corresponding to the absolute pressure in the test section for computing the ΔT.

In the preliminary phase of the investigation it was discovered that the reproducibility of the calorimeters, for determination of quality below the level which could be found by the throttling principle alone, was rather poor. With results taken over a long period of time it was found that the average quality of the steam at the test section, at a pressure of 200 psig, was 94%. Further use of the calorimeters was discontinued, and this value of 94% for quality was used as a basis for all further computations. The quality of the steam, at each point in the test section corresponding to the location of the thermocouples, was computed by adjusting the

enthalpy of the incoming steam in an amount equal to the heat input to the test section. The quality at pressures lower than 200 psig was determined by the use of the Mollier Diagram.

CHAPTER V

CONCLUSIONS AND RESULTS

One hundred runs were made of which forty-six were successful. The remainder of the runs were eliminated because the tube wall temperatures obtained indicated that steady-state conditions had not yet been reached. The ranges of pressure, temperature, quality, and mass rate of flow are given in Table I of Appendix I.

In the process of collecting data a number of experimental errors became apparent. These errors, however, did not exceed the range of expectancy ordinarily associated with this type of investigation. The temperature determinations were expected to be in error at least 2 to 3%. This was not serious as long as the temperature difference between the fluid and the wall surface was large. At small temperature differences care had to be taken to insure that unreasonable results were not obtained. Another error is introduced in the calculation of the transfer coefficient by neglecting axial heat loss from the test section. In estimating this loss it was found to be small, in the range of 10 to 40 BTU/hr. The error due to the radial heat loss was approximately 2% of the input power.

Curves for the heat transfer coefficient versus the percent moisture were made for each mass velocity. In addition a plot was made of the transfer coefficient versus the temperature difference between the wall and the fluid, and one

of the transfer coefficient versus the mass rate of flow. These curves are seen in Figures 6 through 12. The physical properties of steam-water mixtures were not available, thus preventing any correlation of the transfer coefficient with any existing data.

The curves depicting the heat transfer coefficient versus the percent moisture show a very gradual rise in the coefficient up to moisture content between one and two percent. Above this value the coefficient increased rapidly as the moisture increased from two to five percent. A possible explanation of this phenomenon is that at high qualities there are only small droplets of water in the fluid, but not enough to materially disrupt the boundary layer between the steam and the wall of the test section. As the amount of moisture in the steam increases, the water droplets become larger and begin to rupture the boundary layer and collect on the wall of the test section, with a consequent increase in the heat transfer coefficient. The sharp change in slope of the curve is the point at which the water begins to collect in droplets on the tube wall. With a further increase in moisture, there is an approach to annular flow where the boundary layer becomes completely liquid. At this point the transfer coefficient should reach a maximum and then start to decrease. Fully developed annular flow was never reached in this experiment.

Figure 11 shows the influence of mass flow rate on the heat transfer coefficient. It is surprising to note that the coefficient decreased as the flow rate increased to a value of about 200 lbs/hr. With flow rates above this value, the coefficient increased. Only one point on the curve was established beyond this transition region. However, the predicted curve is extrapolated (dotted line) on the plot. The initial decrease in the coefficient was possibly due to the water droplets on the wall being flattened out as the flow rate increased and thereby forming a liquid boundary layer. As the mass flow rate increased beyond 200 lbs/hr the velocity of the gas was apparently sufficient to start wiping the water droplets off the wall instead of just flattening them out. This is how the continued upward trend in the curve was predicted.

Figure 12 indicates curves of the transfer coefficient versus the difference in temperature between the tube wall and the fluid. This shows the correlation between the heat transfer coefficient and temperature difference, for a given range in quality of about five percent. These curves appear to be in the region of unstable film boiling when compared to the boiling curve for free convection(7), but the coefficients determined in the investigation are necessarily higher than those of the boiling curve due to the influence of forced convection.

The available literature and results of this investigation permits the conclusion that a maximum heat transfer coefficient exists with respect to the moisture content of the mixture. This point has not been within the range of the subject investigation. The location of this maximum represents an optimum for future design of certain types of heat transfer equipment.

Additional research would be amply justified in order that the peak in this curve be definitely established.

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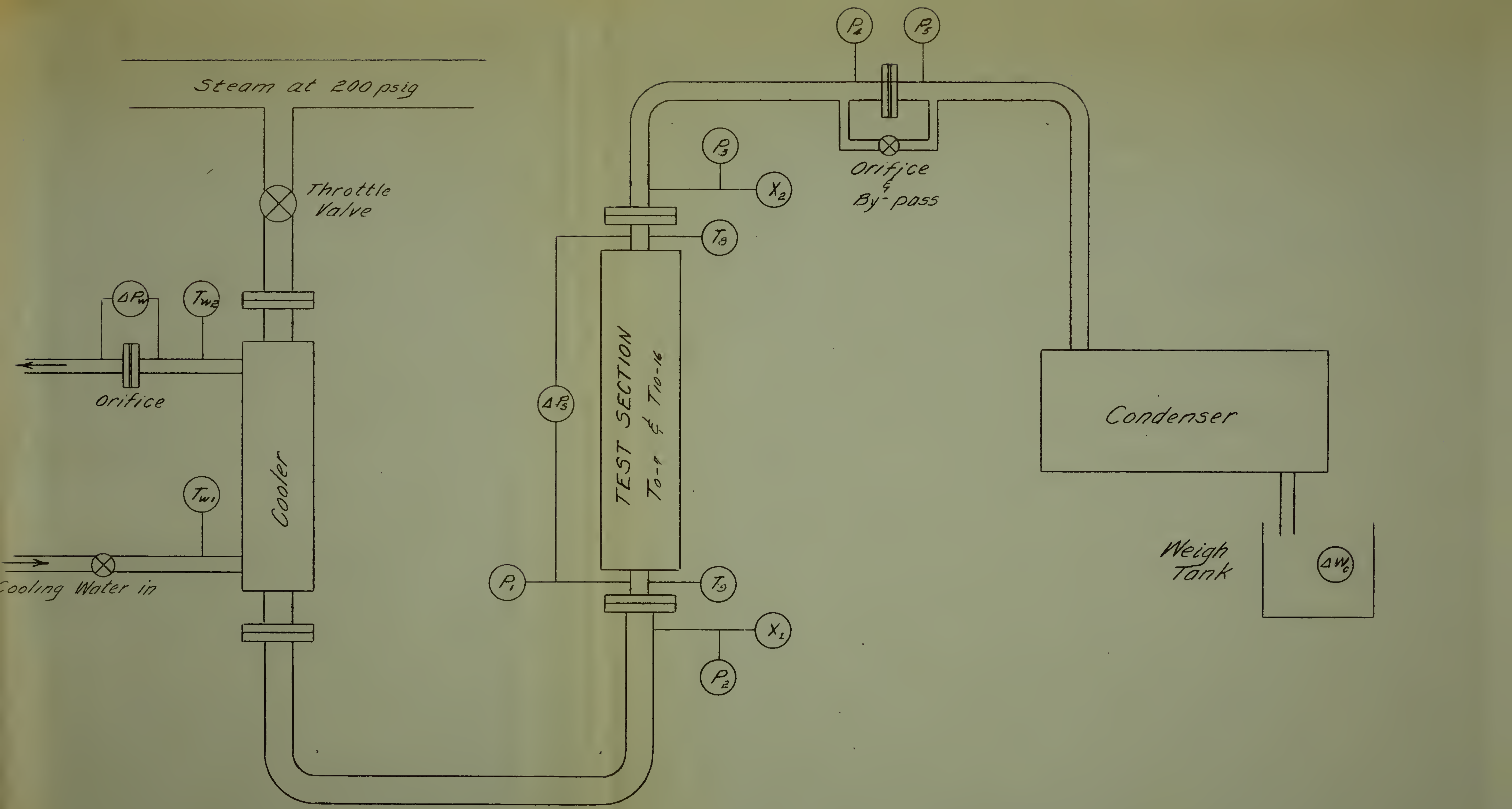
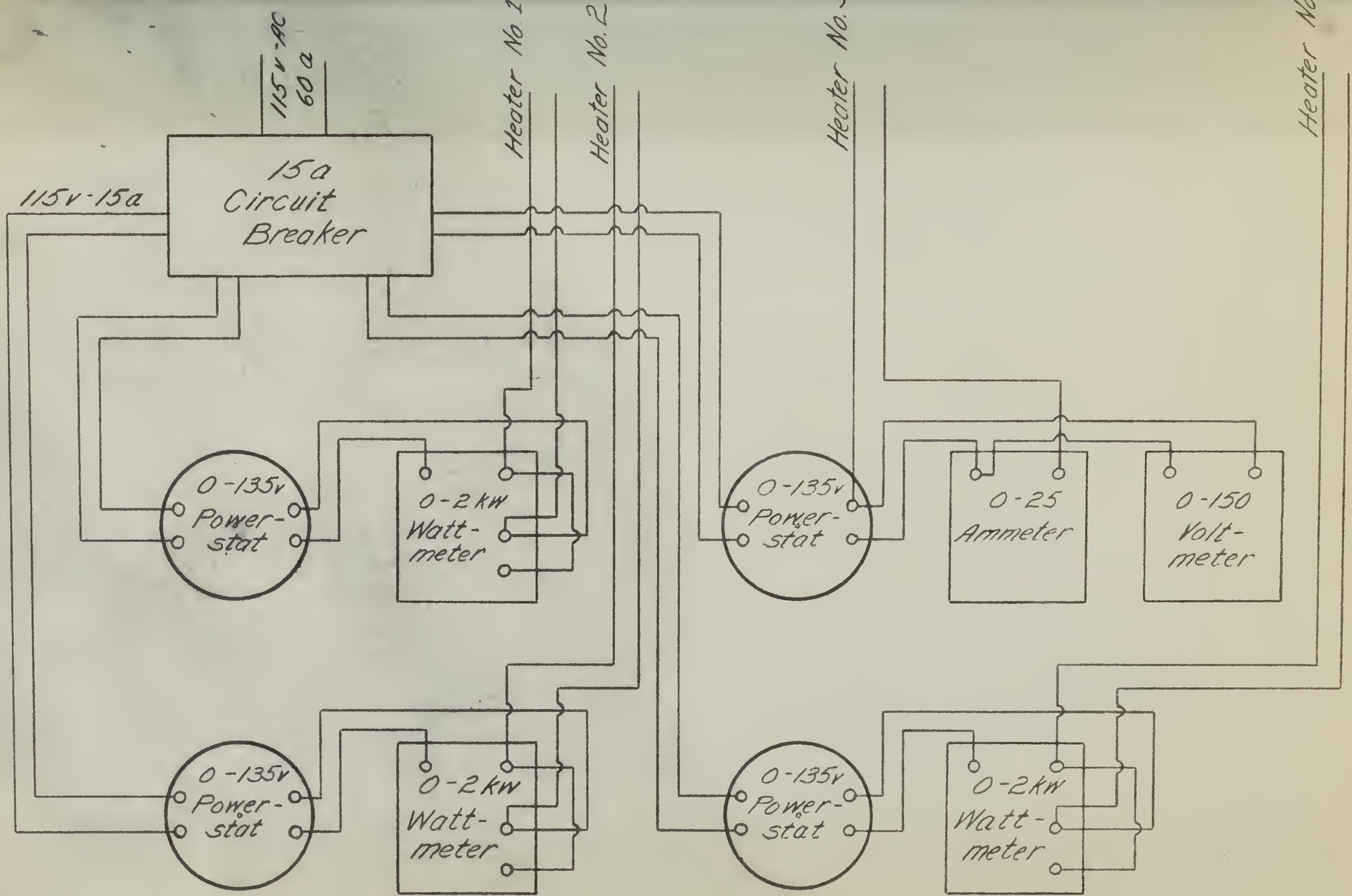


SCHEMATIC - HEAT FLOW APPARATUS - FIGURE 1

Angle θ	C	H
60½°	0.6298"	0.0851"
90°	0.8838"	0.1831"
118°	1.071"	0.3031"

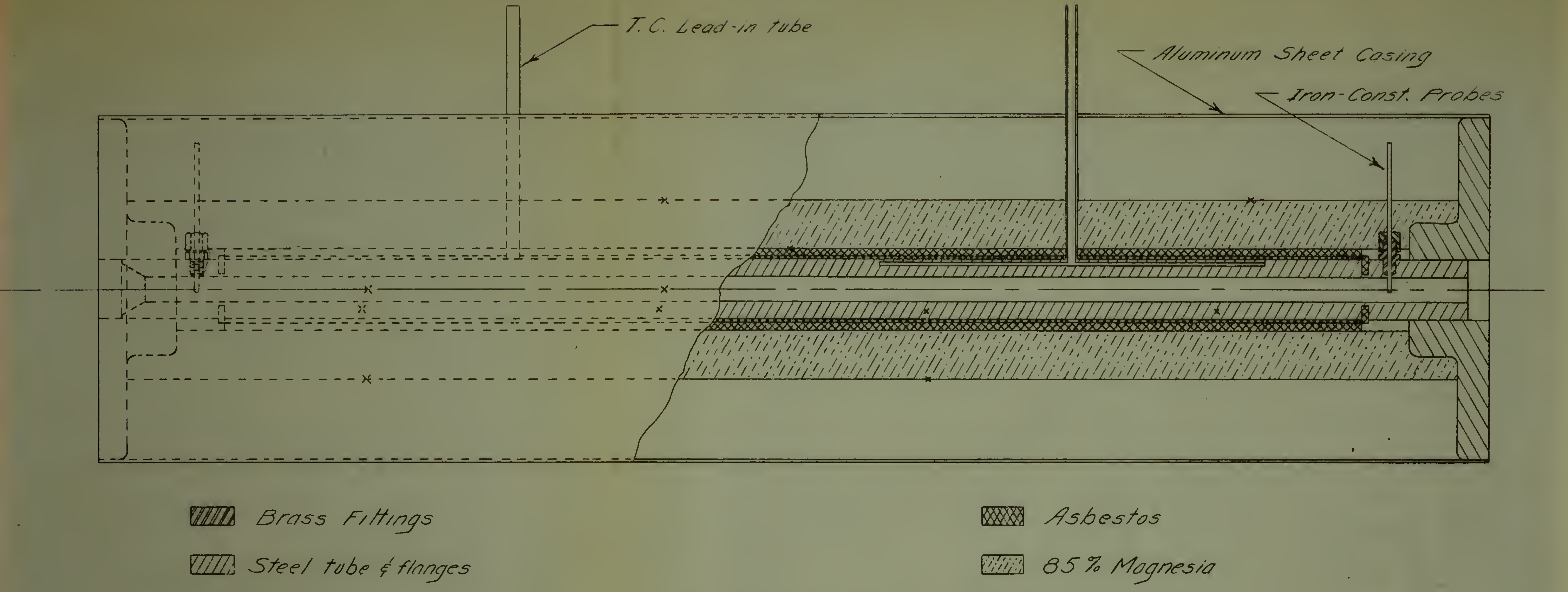
Chord Table

DETAIL of TEST SECTION - FIGURE 2

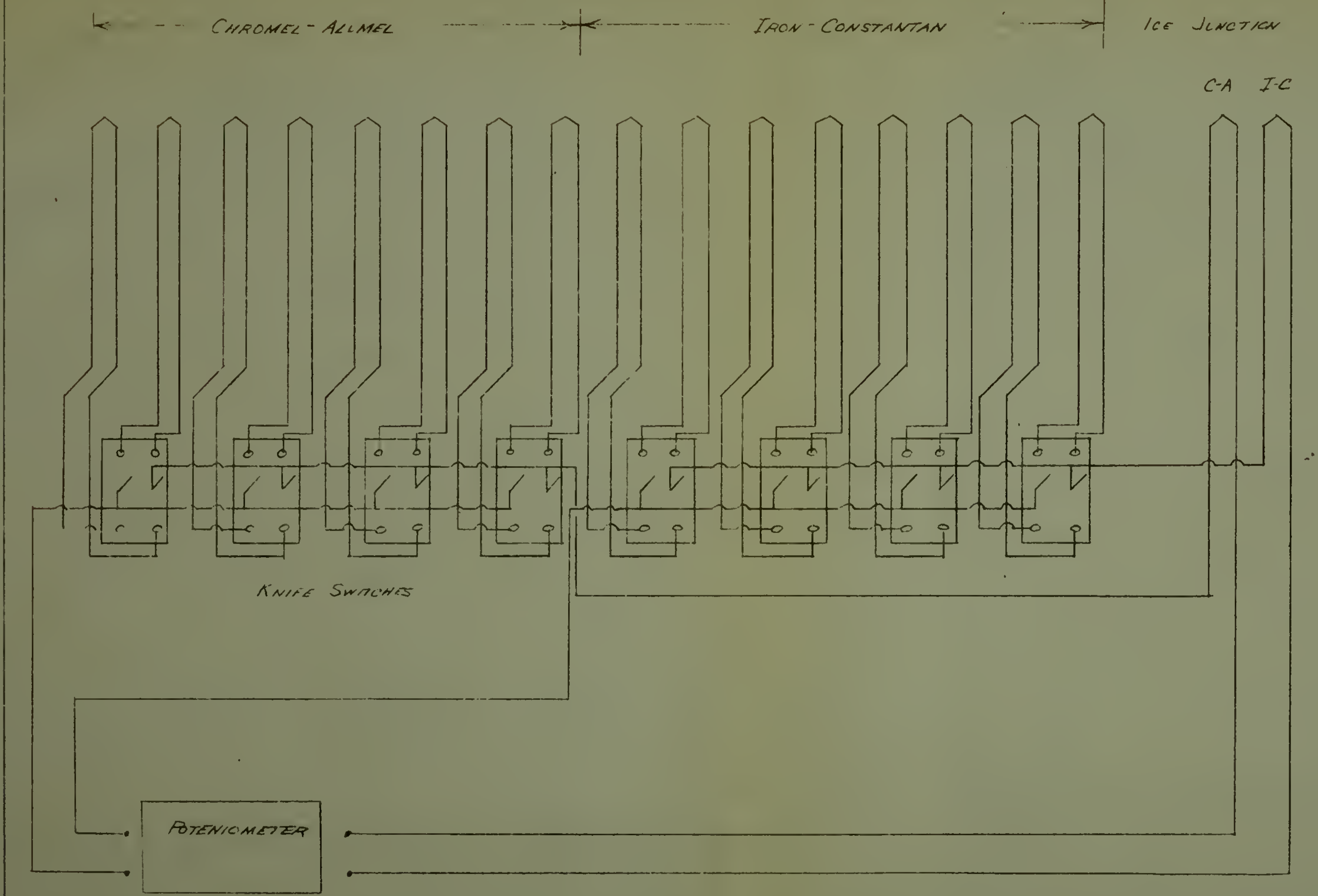


SCHEMATIC of HEATER CONTROL CIRCUIT

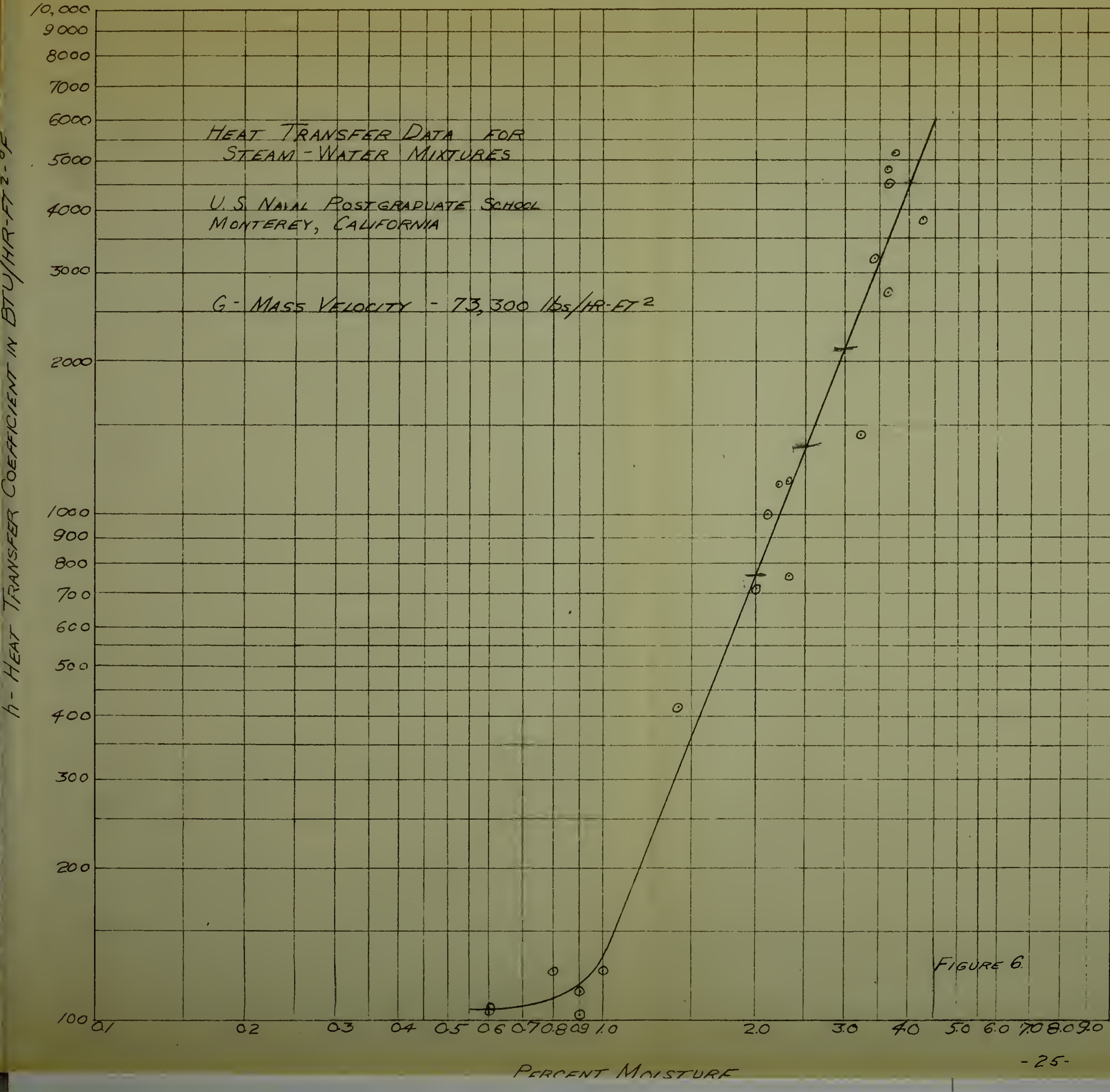
FIGURE 3

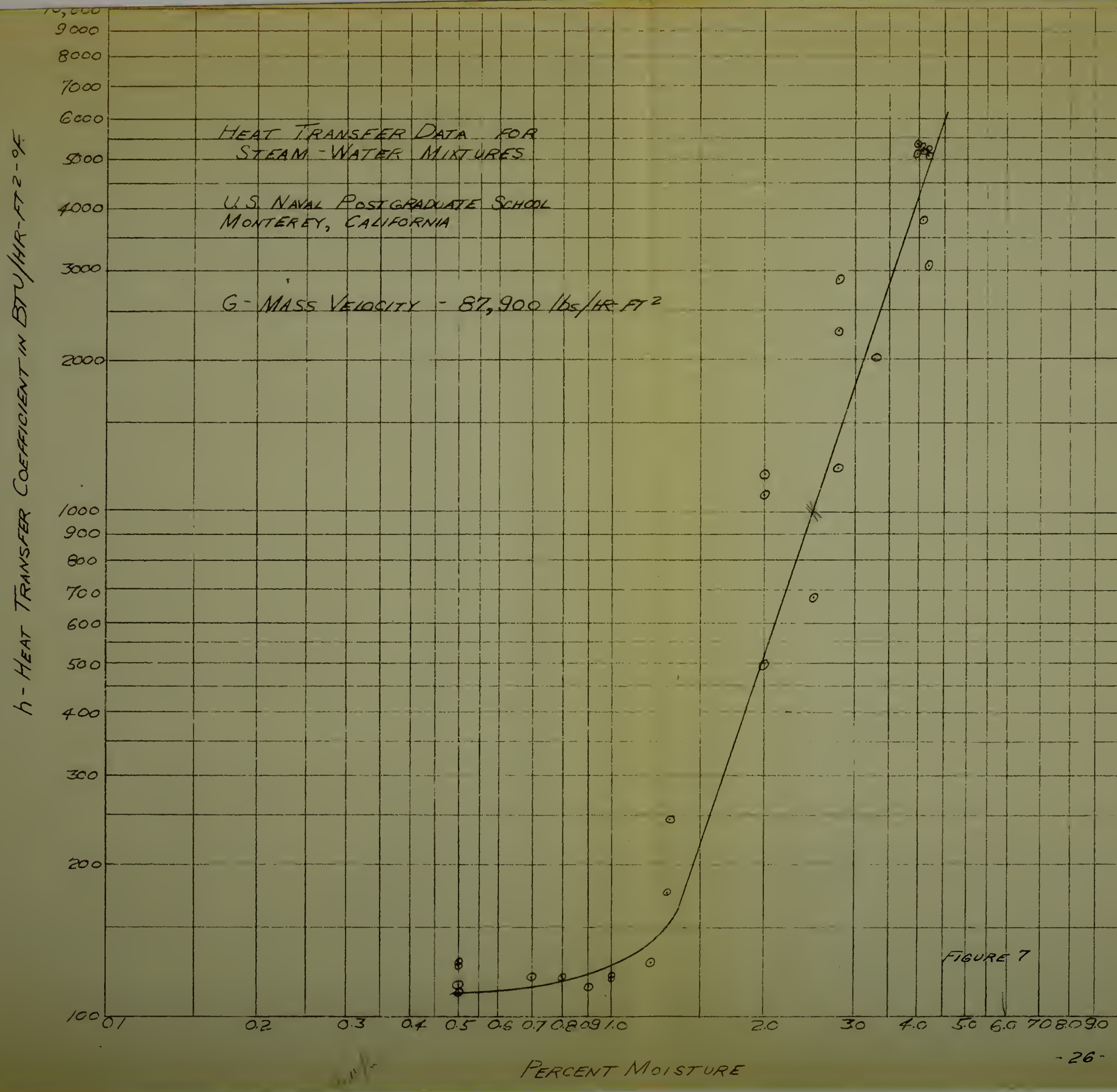


TEST SECTION - FIGURE 4

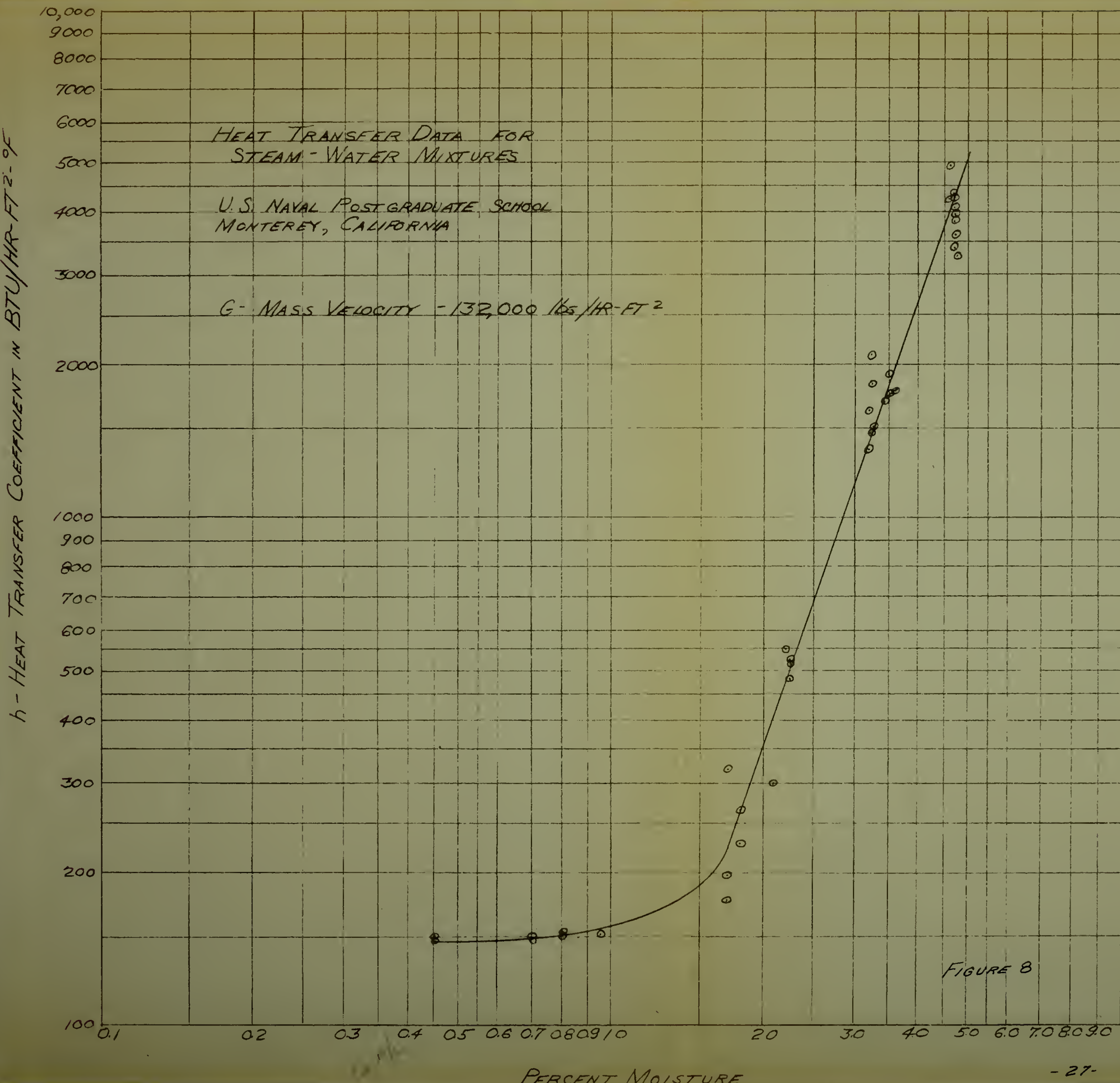


SCHEMATIC - THERMOCOUPLE CIRCUIT - FIGURE 5


HEAT TRANSFER DATA FOR
STEAM-WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
G - MASS VELOCITY - 73,300 lbs/HR-FT²
h - HEAT TRANSFER COEFFICIENT IN BTU/HR-FT²-°F
PERCENT MOISTURE
10,000
9000
8000
7000
6000
5000
4000
3000
2000
1000
900
800
700
600
500
400
300
200
100
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
FIGURE 6



HEAT TRANSFER DATA FOR
STEAM - WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
G - MASS VELOCITY - 87,900 lbs/HR-FT²
h - HEAT TRANSFER COEFFICIENT IN BTU/HR-FT² - °F
9000
8000
7000
6000
5000
4000
3000
2000
1000
900
800
700
600
500
400
300
200
100
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
PERCENT MOISTURE
FIGURE 7



HEAT TRANSFER DATA FOR
STEAM - WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
G - MASS VELOCITY - 132,000 lbs/HR-FT 2
FIGURE 8
h - HEAT TRANSFER COEFFICIENT IN BTU/HR-FT2-°F
PERCENT MOISTURE
10,000
9000
8000
7000
6000
5000
4000
3000
2000
1000
900
800
700
600
500
400
300
200
100
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0

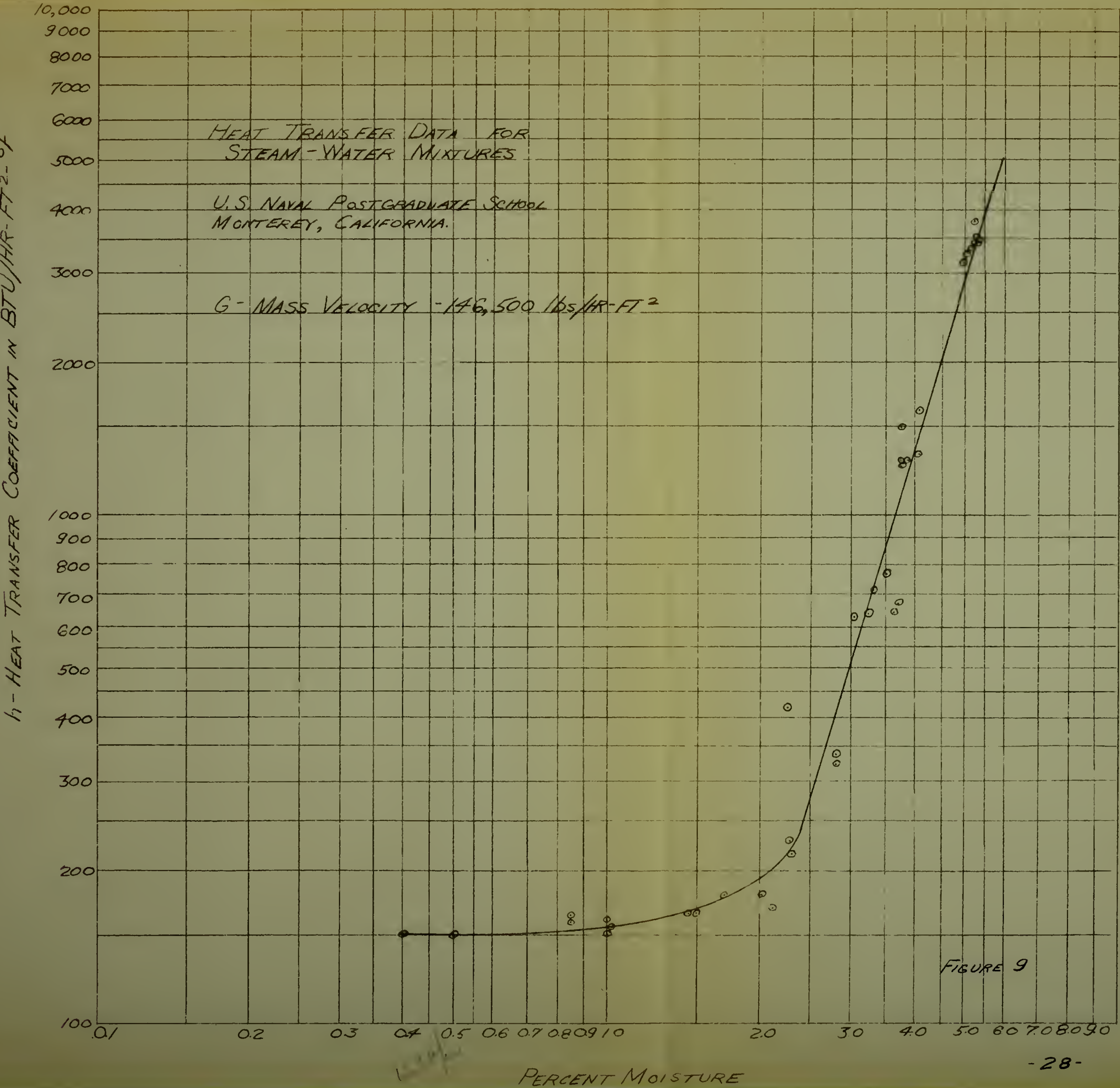
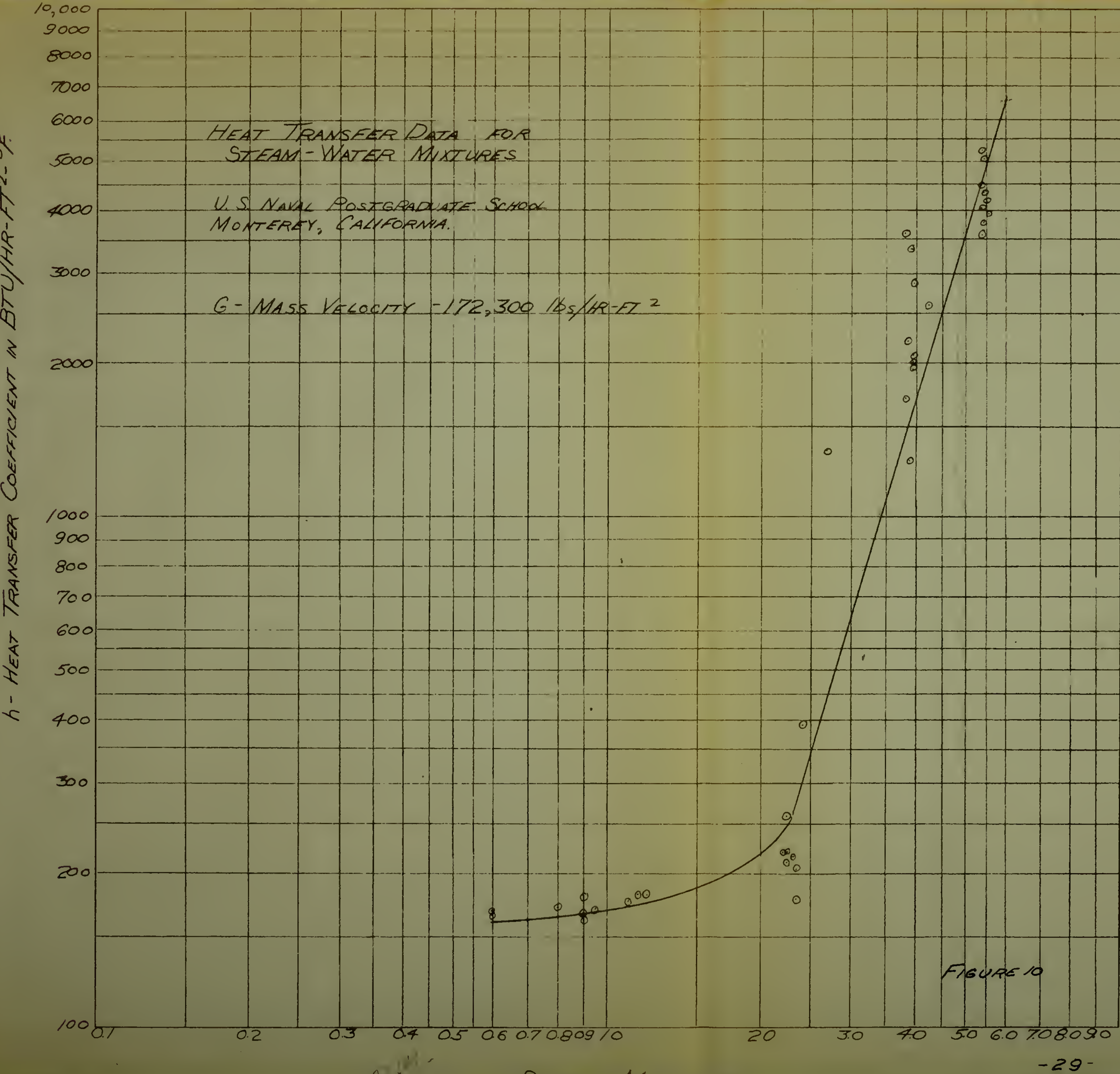
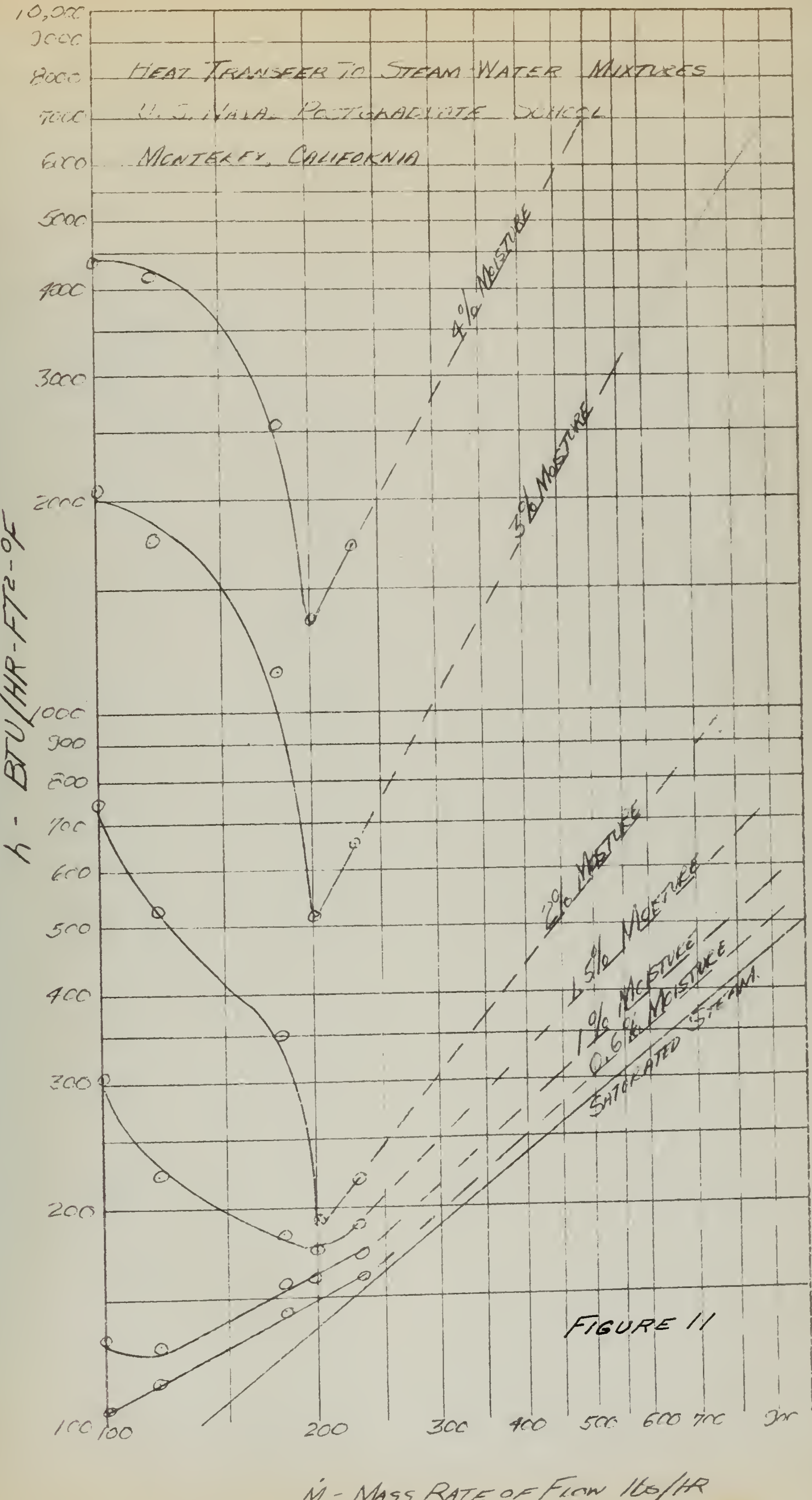

HEAT TRANSFER DATA FOR
STEAM - WATER MIXTURES
U.S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA.
G - MASS VELOCITY - 146,500 lbs/HR-FT²
h - HEAT TRANSFER COEFFICIENT IN BTU/HR-FT²-°F
10,000
9000
8000
7000
6000
5000
4000
3000
2000
1000
900
800
700
600
500
400
300
200
100
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
PERCENT MOISTURE

FIGURE 9



HEAT TRANSFER DATA FOR STEAM-WATER MIXTURES
U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA.
G - MASS VELOCITY - 172,300 lbs/HR-FT 2
h - HEAT TRANSFER COEFFICIENT IN BTU/HR-FT2-°F.
10,000
9000
8000
7000
6000
5000
4000
3000
2000
1000
900
800
700
600
500
400
300
200
100
0.1
0.2
0.3
0.4
0.5
0.6
0.7
0.8
0.9
1.0
2.0
3.0
4.0
5.0
6.0
7.0
8.0
9.0
PERCENT MOISTURE

FIGURE 10



HEAT TRANSFER TO STEAM-WATER MIXTURES
U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA
4% MOISTURE
3% MOISTURE
2% MOISTURE
1.5% MOISTURE
1% MOISTURE
0.6% MOISTURE
SATURATED STEAM
h - BTU/HR-FT²-°F
Ṁ - MASS RATE OF FLOW lbs/HR

FIGURE 11

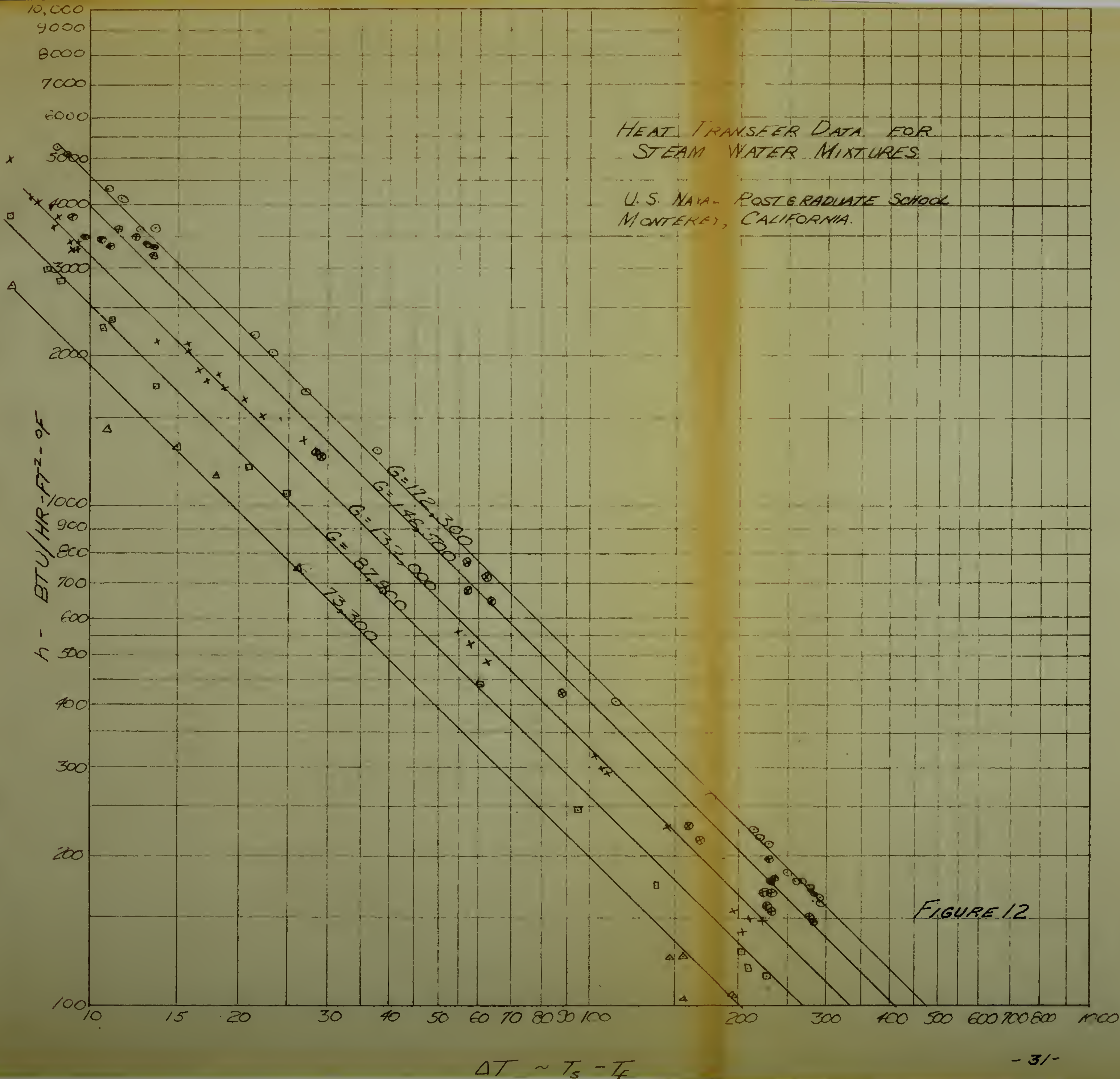

HEAT TRANSFER DATA FOR STEAM WATER MIXTURES
U. S. NAVAL POSTGRADUATE SCHOOL
MONTEREY, CALIFORNIA.
G=172,300
G=146,500
G=132,000
G=87,500
73,300
h - BTU/HR-FT^2-°F
ΔT ~ T_S - T_F

FIGURE 12

APPENDIX

TABLE I

SUMMARIZED DATA

Mass Flow Rate 100 lbs/hr.

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec. Temp °F	T_s	x	h	Q/A
1	103	329.96	377	1	355.7	338.2	.966	3170	26,050
					342.9				
				2	456.1				
					397.4	392.7	.986	415	25,900
				3	633.2				
					606.4	598	Supht	--	25,800
				4	662.9				
					655.6	652.5	Supht	--	26,050
2	97	325.6	351	1	345.0				
					333.3	329.6	.963	5200	20,800
				2	363.5				
					347.3	343.6	.977	1160	20,800
				3	551.5				
					523.5	519.8	.994	107	20,800
				4	577.3				
					574.0	573	Supht	--	20,800
3	98	326.35	351	1	341.5				
					333.3	330.98	.969	4500	20,800
				2	352.5				
					348.8	347.15	.979	1000	20,800
				3	512.5				
					509.0	507	.991	115	20,800
				4	568.5				
					567.5	567	Supht	--	20,800

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec. Temp °F	T_s	x	h	Q/A
4	98	326.35	345	1	341.0				
					334.0	330.5	.964	4840	19,800
				2	355.5				
					345.0	341.5	.978	1310	19,800
				3	510.3				
					485.0	481.3	.992	127.9	19,800
				4	541.0				
					538.2	537	Supht	--	18,150
5	98	326.35	340	1	348.5				
					336.5	333.2	.964	2740	18,150
				2	372.8				
					355.3	352.5	.977	750	18,150
				3	496.5				
					475.5	472.2	.990	125.1	18,150
				4	541.0				
					532.5	530	Supht	--	18,150
6	98	325.6	325.6	1	339.5				
					332.5	329.7	.958	3820	16,200
				2	345.5				
					339.3	336.5	.968	1431	16,200
				3	368.0				
					351.5	348.6	.980	714	16,200
				4	488.3				
					483.8	480.9	.941	103.4	16,200
Mass Flow Rate = 120 lbs/hr.									
7	115	338.1	338.1	1	359.2				
					346.4	342.7	.958	5210	20,800
				2	368.0				
					354.5	348.47	.957	2000	20,800
				3	400.0				
					385.0	379	.980	500	20,800
				4	488.3				
					483.8	480.9	.941	103.4	16,200

[illegible]

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec. Temp °F	T_s	x	h	O/A
8	125	344.3	367	1	372.9				
					355.9	351.2	.959	3800	26,050
				2	404.6				
					387.4	382.7	.925	679	26,200
				3	602.4				
					573.4	569.2	.991	116	26,050
				4	629.6				
					623.6	618.9	Supht	--	26,050
9	128	346.1	363	1	373.2				
					359.4	354.7	.958	3040	25,800
				2	390.6				
					371.4	366.7	.972	1210	25,800
				3	580.2				
					552.9	548.2	.988	129.1	25,800
				4	512.1				
					606.6	601.9	Supht	--	26,050
10	123	343.1	359	1	367.2				
					351.9	347.6	.959	5250	23,600
				2	373.3				
					357.9	353.6	.972	2290	23,650
				3	519.4				
					481.9	477.7	.987	176.5	23,600
				4	577.6				
					569.9	567.9	Supht	--	23,500
11	125	344.3	359	1	365.7				
					353.4	349.1	.958	5070	23,600
				2	372.1				
					356.9	352.6	.972	2860	23,500
				3	488.9				
					443.2	439	.987	247	23,500
				4	572.9				
					564.6	562	Supht	--	23,500

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec Temp °F	T_s	x	h	Q/A
12	126	344.9	345.5	1	372.9				
					355.9	350.3	.960	5140	32,000
				2	400.1				
					379.9	374.3	.980	1095	32,000
				3	629.9				
					598.9	593.3	.995	127.5	32,000
				4	663.9				
					657.3	655	Supht	--	32,000
13	126	344.9	344.9	1	379.1				
					376.5	350.8	.960	5300	31,900
				2	406.1				
					379.1	373.5	.980	1190	31,300
				3	631.9				
					595.6	590	.995	126.9	31,300
				4	667.6				
					661.1	660	Supht	--	31,300
Mass Flow Rate = 180 lbs/hr									
14	192	378.4	395	1	417.4				
					394.4	388	.953	4340	42,700
				2	518.1				
					418.4	412	.970	1240	41,600
				3	667.4				
					635.9	626	.987	169	41,600
				4	685.4				
					681.4	680	Supht	--	41,600
15	187	376.2	381	1	401.54				
					384.54	384.54	.953	4000	33,400
				2	412.34				
					395.34	398.33	.967	1505	33,300
				3	545.04	499.84	.982	270	33,400
					499.84				
				4	611.34				
					602.64	602.64	.995	147.5	33,400

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec Temp °F	T_s	x	h	Q/A
16	187	3762	381	1	406.9				
					389.9	383.84	.954	4380	33,400
				2	418.4				
					402.6	396.54	.968	1645	33,200
				3	564.4				
					528.2	522.14	.982	229	33,400
				4	617.4				
					607.4	601.34	.996	148	33,500
17	186	375.75	380	1	403.9				
					389.9	383.84	.953	4130	33,600
				2	417.7				
					397.6	391.54	.968	2110	33,500
				3	530.4				
					485.9	479.84	.983	320	33,600
				4	611.4				
					604.9	598.84	.995	149.5	33,800
18	187	376.2	380.5	1	404.4				
					390.9	384.84	.953	38[illegible]0	33,600
				2	415.9				
					400.4	394.[illegible]4	.968	1895	33,700
				3	568.7				
					528.2	522.14	.982	229	33,600
				4	618.4				
					609.9	603.84	.996	149	33,800
19	187	376.2	389	1	409.9				
					390.6	384.7	.954	4290	36,400
				2	424.4				
					408.4	402.87	.968	1369	36,400
				3	617.4				
					589.4	583.87	.984	176.2	36,400
				4	637.7				
					651.9	635.37	Supht	--	36,400

[illegible]

Run No.	Press Psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec Temp °F	T_s	x	h	Q/A
20	187	376.2	389	1	410.4				
					390.0	383.5	.954	5000	36,400
				2	426.9				
					407.0	400.5	.968	1495	36,400
				3	611.9				
					568.0	561.5	.983	197	36,400
				4	656.7				
					638.0	631.5		--	36,400
21	188	376.6	376.6	1	408.4				
					391.6	386	.953	3395	31,800
				2	418.4				
					400.9	395.3	.967	1705	31,800
				3	534.4				
					488.6	483	.979	297.5	31,600
				4	603.4				
					594.9	589.3	.993	150	31,800
22	188	376.6	376.6	1	408.9				
					392.9	387.3	.953	2995	31,800
				2	418.1				
					403.9	398.3	.966	1475	31,800
				3	534.9				
					491.9	486.3	.980	291	31,800
				4	603.7				
					595.4	589.8	.993	149.5	31,800
23	188	376.6	376.6	1	408.4				
					391.6	385.74	.953	3320	30,250
				2	416.4				
					399.4	393.54	.965	1777	30,000
				3	494.4				
					439.9	434.04	.978	527	30,250
				4	588.4				
					579.90	574.04	.992	153.5	30,250

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec Temp °F	T_s	x	h	Q/A
24	188	376.6	376.6	1	406.9				
					390.9	385.04	.953	3620	30,500
				2	414.9				
					398.9	393.04	.965	1851	30,350
				3	483.7				
					436.9	431.04	.978	556	30,250
				4	586.4				
					578.7	572.84	.992	154	30,250
25	188	376.6	376.6	1	405.9				
					385.9	385.9	.953	3290	30,500
				2	414.4				
					393.6	393.6	.965	1781	30,100
				3	493.7				
					439.1	439.1	.978	483	30,250
				4	587.9				
					573.1	573.1	.992	153.8	30,250
Mass Flow Rate 200 lbs/hr.									
26	207	384.7	391	1	426.2				
					403.5	396.04	.948	3620	41,300
				2	474.2				
					450.0	442.54	.963	679	40,900
				3	653.5				
					622.3	614.84	.979	171	41,500
				4	681.2				
					670.4	662.94	.995	150	41,550

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec Temp °F	T_s	x	h	Q/A
27	207	384.7	393	1	427.7				
					404.5	397.04	.998	3490	42,400
				2	487.7				
					456.8	449.34	.964	649	42,400
				3	653.2				
					622.8	615.34	.980	181.5	42,400
				4	679.9				
					669.8	662.34	.996	151.3	42,400
28	207	384.7	395	1	427.7				
					405.5	398	.951	3190	42,500
				2	490.2				
					458.5	451	.968	641	42,450
				3	656.5				
					626.3	618.8	.983	181.5	42,450
				4	681.2				
					672.5	665	1.00	151.6	42,500
29	207	384.7	396	1	428.5				
					405.9	398.02	.950	3320	44,200
				2	478.2				
					455.8	447.92	.967	720	44,000
				3	667.2				
					624.8	616.92	.982	197	44,000
				4	696.2				
					685.2	677.32	Supht	--	44,200
30	207	389.7	397	1	428.5				
					405.5	397.7	.949	333.9	43,500
				2	470.2				
					449.5	441.7	.965	775	44,300
				3	664.8				
					629.5	621.7	.983	182	43,200
				4	694.2				
					683.0	675.2	Supht	--	43,500

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Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test Sec Temp °F	T_s	x	h	Q/A
31	207	384.7	384.7	1	418.9				
					402.2	395.67	.948	3330	36,600
				2	416.0	409.47	.963	1505	37,350
					431.7				
				3	538.5				
					478.5	471.97	.978	422	36,600
				4	624.2				
					613.0	606.47	.992	168.5	36,600
32	207	384.7	384.7	1	420.2				
					402.2	395.67	.948	3320	36,500
				2	435.7				
					420.5	413.97	.963	1260	36,950
				3	584.2				
					548.1	541.57	.978	231	36,200
				4	628.9				
					618.5	611.97	.992	160.5	36,500
33	207	384.7	384.7	1	419.2				
					401.8	395.27	.948	3420	36,300
				2	440.2				
					420.0	413.47	.963	1280	36,800
				3	588.0				
					556.8	550.27	.977	217.5	36,000
				4	628.9				
					620.5	613.97	.990	157.1	36,000
34	207	384.7	384.7	1	418.9				
					400.5	394.0	.948	3820	35,800
				2	433.9				
					419.5	413.0	.962	1290	36,500
				3	575.5				
					535.8	529.3	.976	247	35,800
				4	625.2				
					614.5	608.0	.990	161.5	36,100

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sta.	Sec No.	Test sec Temp °F	T_s	x	h	Q/A
35	206	384.3	384.3		1	417.2				
						400.0	393.94	.948	3450	33,350
					2	429.2				
						415.5	409.44	.960	1339	33,550
					3	525.5				
						485.3	479.24	.972	350	33,350
					4	599.7				
						590.8	584.74	.985	166.1	33,350
36	207	384.7	384.7		1	417.2				
						400.5	394.44	.948	3450	33,600
					2	428.5				
						411.5	405.44	.960	1630	33,800
					3	525.5				
						488.1	482.04	.972	343	33,350
					4	601.7				
						593.8	587.74	.986	166.7	33,600
Mass Flow Rate-230 lbs/hr.										
37	212	387.1	391		1	432.2				
						409.0	400.5	.947	3620	47,400
					2	468.2				
						433.0	424.5	.961	1295	47,300
					3	659.8				
						626.3	617.8	.978	210	47,400
					4	686.9				
						686.8	678.3	.994	166.5	47,400
38	212	387.7	391		1	431.2				
						408.0	399.51	.946	3810	47,400
					2	445.7				
						419.5	411.01	.961	1875	47,300
					3	661.5				
						627.3	618.81	.977	204.5	47,400
					4	687.5				
						676.5	668.01	.991	168.5	47,400

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test sec Temp °F	T_s	x	h	Q/A
39	212	387.7	391	1	429.7				
					406.5	398.01	.946	4340	47,400
				2	445.2				
					417.5	409.01	.961	2160	47,300
				3	674.8				
					663.8	655.31	.977	177	47,400
				4	697.2				
					687.5	679.01	.991	163	47,400
40	212	387.7	392	1	428.2				
					406.5	398.01	.946	4370	47,950
				2	442.2				
					418.9	410.41	.961	2050	47,950
				3	651.5				
					615.1	606.61	.977	218	47,950
				4	684.5				
					672.5	664.01	.992	173	47,950
41	212	387.1	392	1	432.7				
					407.2	398.71	.947	4080	47,400
				2	442.7				
					423.2	414.71	.962	1716	47,400
				3	648.3				
					610.1	601.61	.978	223	47,400
				4	687.5				
					676.5	668.01	.994	168.5	47,400
42	212	387.1	392	1	430.5				
					405.0	396.5	.946	5050	47,400
				2	445.5				
					417.0	408.5	.962	2220	47,400
				3	649.5				
					610.1	601.6	.978	222	47,750
				4	684.2				
					674.2	665.7	.991	171.5	47,750

Run No.	Press psia	Inlet Temp °F	Outlet Temp °F	Test Sec Sta. No.	Test sec Temp °F	T_s	x	h	Q/A
43	212	387.1	387.1	1	426.7				
					405.5	397.29	.947	4500	45,800
				2	432.2				
					408.8	400.59	.961	3400	45,600
				3	577.0				
					509.3	501.09	.978	402	45,600
				4	651.2				
					645.2	636.99	.989	183.8	45,400
44	212	387.1	387.1	1	422.2				
					404.0	395.79	.947	5270	45,800
				2	433.5				
					408.0	399.79	.962	3600	45,800
				3	605.0				
					570.5	562.29	.978	260	45,800
				4	661.2				
					653.8	645.59	.989	176.5	45,600
45	212	387.1	387.1	1	424.7				
					405.0	397.17	.946	4240	43,700
				2	432.2				
					410.0	402.17	.961	2880	43,500
				3	578.8				
					507.8	499.97	.976	390	43,700
				4	645.2				
					634.0	626.17	.991	183	43,700
46	212	387.1	387.1	1	427.2				
					406.0	398.17	.945	3960	43,700
				2	432.2				
					411.5	403.67	.958	2620	43,600
				3	512.0				
					437.3	429.47	.973	1035	43,700
				4	646.2				
					634.8	626.97	.988	182.5	43,700

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TABLE II

THERMOCOUPLE ERROR - ^{o}F

Press psia	1	2	3	4	5	6	7	8	9	10	11	12
199.5	-1.2	-1.5	-1.5	-10.8	-1.2	-6.5	-1.2	-4.5	-8.2	-4.8	-8.8	-8.2
177.5	-1.4	-1.4	-1.4	-8.9	-0.4	-4.9	-0.9	-3.9	-7.1	-4.4	-7.7	-6.4
157	-0.3	-0.8	-0.8	-8.3	-0.3	-4.8	-0.2	-2.0	-6.3	-3.6	-7.3	-6.3
128	-0.5	-0.5	-0.8	-8.0	-0.0	-9.5	-0.5	-5.0	-5.5	-3.5	-6.2	-5.5
100	-0.6	-0.6	-0.9	-5.9	-0.6	-2.4	-0.2	-2.4	-5.9	-3.9	-6.2	-5.6
75	-0.0	-0.5	-1.0	-5.0	-0.5	-2.3	-0.5	-1.0	-5.3	-2.3	-5.3	-4.7
49	-0.6	-0.6	-1.1	-5.1	-1.1	-2.1	-0.6	-2.6	-5.9	-3.9	-5.9	-4.6
35.6	-1.9	-2.4	-2.4	-4.1	-1.9	-3.7	-1.4	-1.9	-6.7	-3.7	-6.1	-5.4
19.0	-2.0	-1.5	-2.0	-3.0	-1.5	-2.0	-1.5	-1.5	-6.0	-4.3	-5.7	-5.3

[illegible]

[illegible]

[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

TABLE III

CALCULATED TEMPERATURE DROP FOR VARIOUS POWER INPUTS

Power Input per Coil in Kilowatts	Temperature Difference Between Surface Thermocouple and Inside Surface	Temperature Difference Between Depth Couple and Inside Surface
300	11.94	2.80
350	13.92	3.26
400	15.91	3.73
450	17.90	4.20
500	19.92	4.66
550	21.9	5.13
600	23.9	5.60
650	25.9	6.06
700	27.9	6.53
750	29.8	u.00
800	31.8	u.46
850	33.8	7.92
900	35.8	8.40
950	37.8	8.86
1000	39.8	9.32

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